A COMBINED 1D-3D MODELING APPROACH FOR THE DESIGN AND ANALYSIS OF COMPLEX MECHATRONIC SYSTEM

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ABSTRACT

Nowadays, virtual prototyping is often incorporated in the design process to accelerate the development process of complex mechatronic systems [1]. This implies that the use of experimental campaigns has to be reduced and the manufacturer has to rely more on simulation tools [2]. In this paper, the on-going activities on the simulation and validation of combined 1D and 3D models, for the design and analysis of complex mechatronic system, have been presented. This includes the development of a flexible multibody model, a lumped parameter driveline model and a control system. In order to exhibit the potential of the virtual design and analysis process for modern mechatronic systems, an industrial machine tool is used as a case study. For predicting the dynamic behavior of the machine, forecasting the influence of specific design changes, and assessing the impact of different control architectures with full confidence, the model needs to be validated. To this end, the simulation results of the virtual model is compared with the results obtained on the physical prototype.

KEY WORDS
Mechatronic system modeling, Virtual prototyping

1 Introduction

The rising demand of high speed and high precision mechatronic systems, while reducing the time to market, motivates to include virtual prototyping in the design and development process [1, 3, 4, 5]. Examples of such mechatronic systems include pick-and-place machines [6], milling machines [5], water jet cutting machines [4], weaving looms [7], 3D rapid prototyping machines, cartesian mechanisms etc. Such a system consists of several sub-systems or modules from different engineering disciplines varying from hydraulic components over controller hard- and software till electro-mechanical drivelines and storage elements. As each of these different modules interacts with each other, a purely experimental approach is no longer sufficient to characterize and understand the physical behavior of the machinery. In addition, it is crucial to cut the number of design and development cycles while reaching the desired market specifications [3]. Therefore, an integrated strategy has been presented for the simulation and validation of mechatronic system models. A 3-axes machine tool is used to demonstrate the strategy.

1.1 System description

The industrial case study is a 3-axes machine tool, shown in Figure 1. The gantry of the machine tool moves in the X-direction, whereas the head is capable of moving along the Y and Z directions. As the variation in the Z-axis is expected to be small during machine operation and has a negligible influence on the dynamic behavior , it is not taken into account for the analysis. The total mass of the gantry is 330 kg including the head mass of 50kg. The machine is equipped with two rotary motors, along the sides of the gantry, which drive the machine in X-direction via rack and pinion mechanisms. In addtion, a linear motor is used to actuate the machine head in Y-direction. The displacement of the rotary motors are measured via built-in encoders, whereas the position of the linear motor is measured by an optical encoder attached along the length of the gantry. The purpose of this machine is to move the tool center point (TCP), fixed on the machine head, along a given trajectory in the workspace as fast and precisely as possible. An experimental prototype of the machine tool is shown in Figure 2.

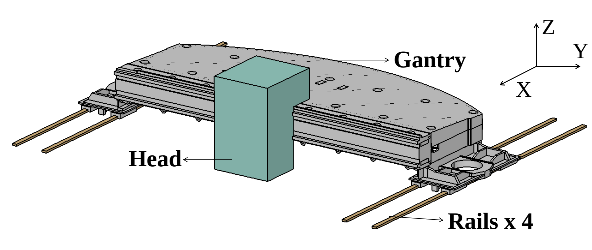

Figure 1. CAD model of an industrial machine tool

Figure 2. Prototype of an industrial machine tool

1.2 Integrated Simulation Strategy- An Overview

In an integrated simulation approach, each module is described separately in their most suitable formalism. The formalism in which these different laws are described depends on the complexity of the system and the desired accuracy; for the machines having elastic components and subjected to significant excitations, flexible multi-body models are required [8]. In contrast, an electric motor or a gear-box can, in some cases, be described by ideal lumped components. These modules or sub-systems are combined with eachother via a bondgraph approach. A more detailed description is given in [7, 9]. After building the plant model, a controller has to be designed based on a low-order model of the system. For the presented case study, it is expected that the gantry of the machine tool will undergo linear elastic deformation during operation; it has to be modeled by a 3D modeling approach that intrinsically takes the exact geometry into account. For describing the mechanical behavior, a flexible multi-body model is required. The rest of the drivetrain (i.e. rack and pinion mechanisms, gear-boxes and motors) can be described by lumped ideal components or even neglected if their behavior has no significant influence on their system level dynamic behavior.

The coupling of such sub-systems in an integrated approach allows us to test different control strategies [6], to evaluate the performance and robustness of the closed loop system [10], to analyze the impact of specific design changes and to assess the performance of reduced order model as well as reduced order controllers. Moreover, if a prototype of the system is available, this methodology allows us to use simulation results to prepare experiments. The obtained experimental results can then be used to update the model. The proposed methodology is shown in Figure 3.

1.3 Paper Outline

In this paper, a 1D-3D combined modeling approach has been demonstrated for an industrial gantry machine. To this end, Section 2 describes the mechatronic system modeling in detail. In order to validate the obtained model, Section 3 discusses the experiments performed on the physical prototype. The measurement setup required to perform these experiments and the adopted procedure is presented. The paper ends with the concluding remarks presented in the Section 4.

2 Mechatronic System Modelling

The complete mechatronic model combines a 3D model (i.e. flexible multi-body model of the gantry), a 1D lumped parameter model and a controller. This Section deals with these different sub-systems of the simulation model independently and discusses how to combine them in an integrated architecture.

2.1 3D Flexible Multi-body model

The flexible multi-body (FMB) model comprises of rigid bodies, flexible bodies, joints, constraints, sensor and actuators. This implies that FMB model requires the models of flexible and rigid bodies, and the mechanisms for the interconnecting rigid and flexible bodies. In addition, a connection with the rest of the system (i.e. 1D model and controller) needs to be established for combining the FMB model with the 1D model. Moreover, a reduced order model is required for efficient simulation purposes. In the sequel, these steps are described in chronological order.

For the current case study, it is assumed that the head of the machine tool is a rigid body. The gantry will undergo elastic deformation during machine operation, in addition to the rigid body motion. Thus, the flexible multi-body model of the machine tool consists of a rigid head and a flexible gantry. Building a flexible body/asssembly starts with creating the finite element (FE) meshes of the individual parts of the assembly. As a finite element mesh is based on the actual geometry, a computer aided design (CAD) model is used as a starting point for creating the mesh. In order to develop the physical prototype, shown in Figure 2, the designers usually develop a detailed CAD model of the machine tool with all the components and auxiliary systems. In practice, every detail of the geometry is not required to take into account. Small auxiliary systems are neglected or assumed rigid to decrease the degrees of freedom (DOF) in the FE mesh. Other details like fillets, chamfers, small holes, grooves etc. are removed if they have a negligible effect on the mode shapes and eigenfrequencies of the individual body [2]. The next step is to combine the different FE meshes of the components together to form an assembly. The complete gantry has 5 components that are connected to each other via bolted connections. Three of these are made up of aluminum and the other two are made up of steel. Each bolted connection is represented by a multiple point constraint (RBE2 element) as described in [11]. The auxiliary systems (i.e. motors, cables, bellows, valves

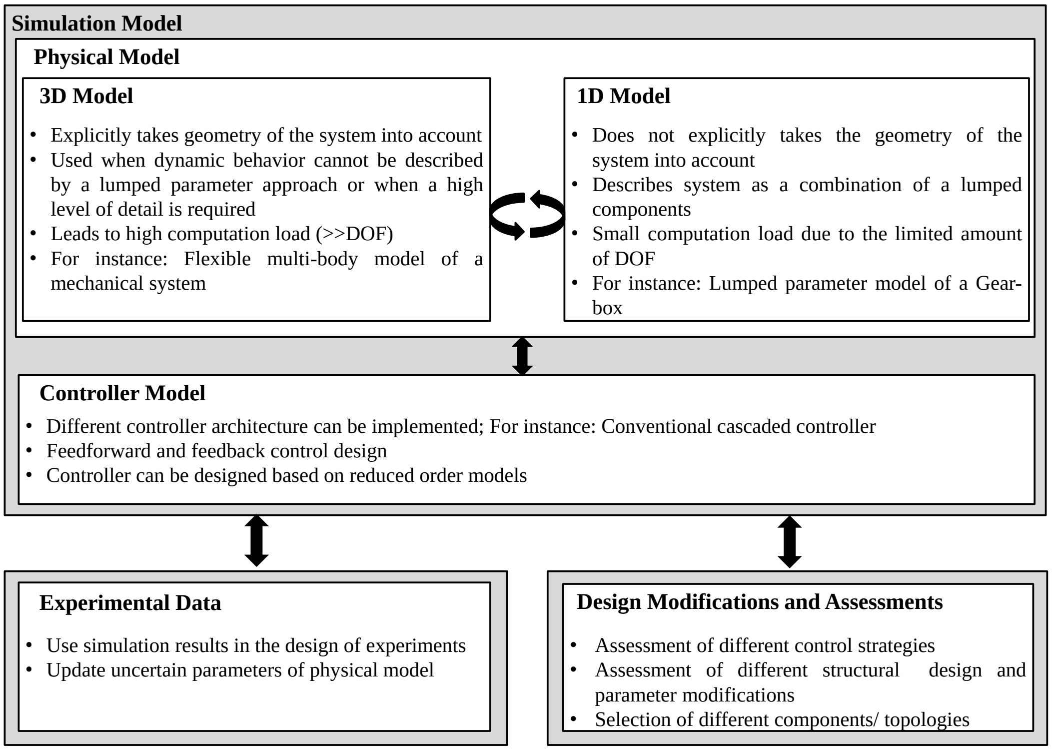

Figure 3. Integrated Simulation Strategy

etc.) are added to the finite element model as an equivalent point mass with the same inertial properties. Similar multiple point constraints (RBE3 element) are used to attach them to the finite element mesh. The connection between the gantry and the guides are defined by a 6-DOF stiffness relation (CELAS element), where the stiffness is set to zero in the translational direction. The model , shown in Figure 4, has been meshed sufficiently dense to ensure convergence. The complete gantry model has approximately 0.7 million degrees of freedom.

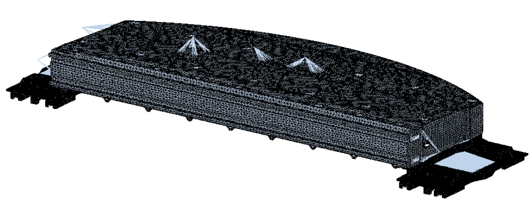

Figure 4. Finite element model of the gantry machine

The model parameters are not always known beforehand and are uncertain or vary over time. This makes it very difficult to correlate the model with reality as the number of uncertain parameters is substantial and thus creating a vast space of possible parameter combinations. The connection between the gantry and the guides are defined by the stiffness values in 5 directions (1 translational direction is left free). A limited set of sensitivity analyses have been performed for different values of stiffness in the guides, and from that it can be conclude that the dynamic behavior is highly dependent on these flexibilities. For the current simulation model, the recommended stiffness values by the machine tool manufacturer are used. These values correspond to one-fourth of the stiffness values given in the datasheet of the guides. The actual values for the stiffness are still uncertain and depend on preload, manufacturing tolerances, lubrication, assembly alignments, etc.

Next, a mechanism can be built by defining joints and constraints between the different bodies at the interface points. All joints are assumed to be ideal and have no flexibilities and friction.The translational joints are used between guides and rails, and flex-point curve joint (see details in [12]) is used to attach the rigid head with the flexible gantry. The flexibilities in the guides for the X-axis are already incorporated in the finite element model, whereas the guides for the Y-axis are assumed to be rigid. Moreover, it is assumed that the rails for the X-axis guides are rigid and attached rigidly with the ground.

When the flexible multi-body model is created, an interface with the rest of the system (i.e. 1D model) is established via control nodes. These control nodes can be used to apply forces and measure displacement, velocity and acceleration. The flexible multi-body model is developed in

LMS Virtual.Lab Motion Environment, shown in Figure 5.

As mentioned earlier, the finite element model for the gantry has approximately 0.7 million DOF; this model is not directly suitable for efficient computer simulation purposes. Therefore, there is a need to obtain a reduced order model. A reduced order model is computed by using the component mode synthesis (CMS) technique [13]- a well-known method in linear structural dynamics. Craig-Brampton modes are computed for the gantry (without head) in a solver package i.e. MSC/MD NASTRAN. With this method, each mode of the flexible body adds one generalized coordinate to the system [12]. The interface or connecting degrees of freedom are preserved [14]. The total number of modes used for the simulation is equals to 60. The obtained reduced order model via CMS is suitable for simulation purposes. Finally, it is necessary to mention that 2.5 % modal damping is used. This estimate is based on the experimental modal analysis performed earlier on the physical prototype.

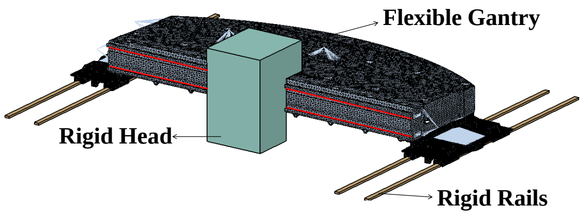

Figure 5. Flexible multi-body model (LMS Virtual.Lab Motion Environment)

2.2 1D Multi-physics model

The other parts of the system, that are relevant for modeling the dynamic behavior but do not require a detailed description of the geometry, are modeled by a lumped parameter approach using the bondgraph method. Although the mechanical components are described only for this particular exercise, this is not a constraint on the methodology. The bondgraph method couples components together by means of energy relations which are independent of their physical domain. This means that every interface point consist of effort and flow variables that uniquely define the power at that particular interface point [7, 9]. This approach of energy relations can be exploited to integrate systems with different formalism together i.e. lumped parameter models can be combined with flexible multi-body models as long as an appropriate energy relationship is defined at the interface points. LMS Imagine.Lab AMESim provides a platform for modeling and analysis of physical multi-domain systems, governed by ordinary differential equation ODE or differential algebraic equations DAE [15]. This platform is used to model the lumped parameter driveline model. The modeled driveline for the X-axis consists of a motor inertia, a gear-box (modeled as an ideal reducer), a rack and pinion mechanism (modeled as an ideal transformer), and the lumped stiffness and damping in the driveline. The modeled drive-line for the Y-axis consists of a linear force input. The stiffness and damping values of the X-driveline is provided by the machine manufacturer. All the other parameters are taken from the datasheets of the components. The models developed in AMESim for both the axes are shown in Figure 6.

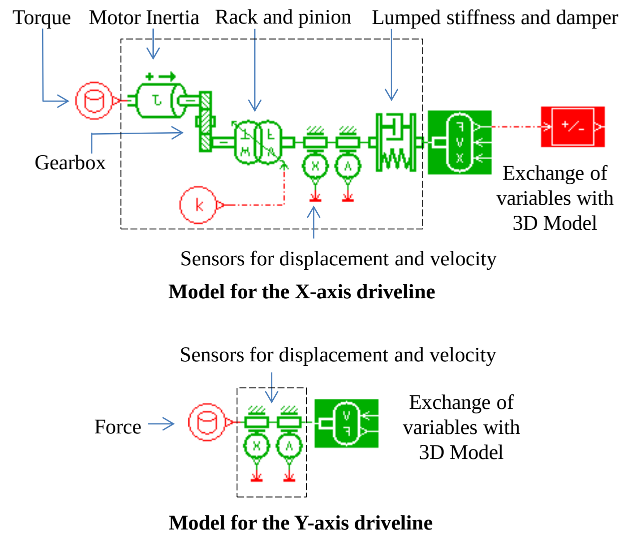

Figure 6. One-dimensional lumped parameter driveline models for the X and Y axes (LMS Imagine.Lab Environment)

2.3 Controller

The accuracy of the machine operation is significantly dependent on the performance of the servo drive's controller. The purpose of the machine is to follow the desired trajectories as quickly and precisely as possible. This implies that the machine should follow geometric trajectories time optimally with limits on the deviation of TCP from the given trajectories. In machine industry, this deviation is referred to as contouring error i.e. the component of error perpendicular to the given trajectory [16]. A cascaded scheme with the P (proportional) and PI (proportional-integral) controllers for the position and velocity loops, respectively, together with velocity and acceleration feedforward has been chosen, as shown in Figure 7. More on servo drive control for machine tools can be found in [16]. There are two main reasons for chosing this type of control scheme: (i) this conventional cascaded control is very common in machine industry, (ii) at present, only this scheme can be implemented on the physical prototype. As a result, a tuned controller for the physical prototype, implemented on the B&R Automation studio platform, can also be validated on the virtual prototype in simulation. The controller for the X and Y axes drives are attached to the corresponding 1D drive-line models in AMESim.

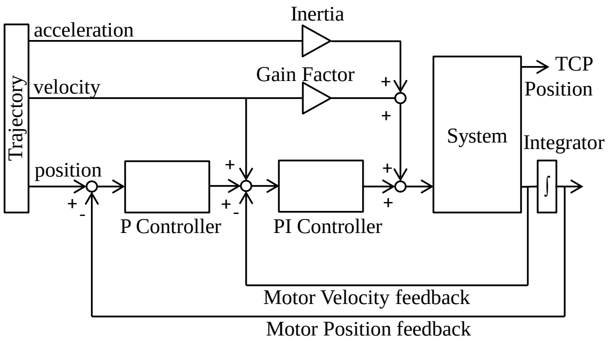

Figure 7. Schematic of cascaded controller

2.4 Model Integration

The 1D lumped parameter model and the 3D flexible multi-body model are built separately on different platforms. In order to the overall dynamic behaviour of the system, these models have to be simulated in an integrated fashion. To accomplish this, the following two approaches are supported by Imagine.Lab AMESim and Virtual.Lab Motion platforms:

- **Co-simulation:** With co-simulation, the state equations for the different components (1D/3D) of the systems are solved independently and their data is exchanged at discrete time steps.

- **Coupled simulation:** In this case, the complete set of state equations for all the components is processed with a master solver.

In each case, the different sub-systems have to be treated as an equivalent bondgraph component in order to interface them with each other. It is chosen to treat the flexible multi-body model as an equivalent inertia.Treating the flexible multi-body model as an equivalent inertia means that it receives an input force on the interface nodes while it gives the displacement and velocity as outputs at the interface nodes. In this particular case, the most straightforward option is to combine both sub-systems via co-simulation, as shown in Figure 8. This approach is justified as long the communication interval, between the two platforms, is small enough inorder to ensure that fast dynamics are not missed by the solver. A communication interval of $100\mu s$ has been chosen for the simulation; however, the data is sampled at every $400\mu s$ to remain consistent with the experimental setup. This also keeps the size of the logged output file comparatively smaller.

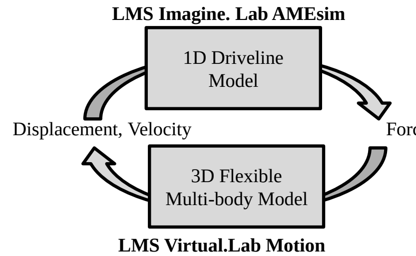

Figure 8. Co-simulation between LMS Virtual.Lab and LMS Imagine.AMEsim

3 Experimental and virtual identification

Now, the developed mechatronic model is ready for the analysis via co-simulation. This implies that the dynamic behavior of the system can be identified and validated. Since, the physical prototype of the machine tool is available, same experiments can be performed on the physical and virtual prototype. The obtained results can be compared to check the accuracy of the model. In this Section, first, the measurement setup for performing the experiments on the physical prototype is discussed. Next, the technique used to identify the dynamic behavior of the physical and virtual machine is described. In the end, the obtained results are presented followed by the discussion on the results and possible future work.

The reponse of the motors displacements are recorded by synchronously logging the motor encoder signal. The heidenhain KGM grid encoder is used to measure the response at the TCP. The KGM sensor system comprises a scanning head and a grid plate embedded on a base plate. The advantage of this system is to do non-contact displacement measurement of TCP. In order to perform these measurements, the base plate is mounted on the table of the machine tool and aligned by using a dial indicator. Then, the scanning head is attached to the machine head by using a sheet metal bracket fixture.

Often, open loop identification methods are not safe to use on actual physical prototype. For this reason, a closed loop frequency domain identification technique is used to identify the machine tool experimentally, see [17]. The same technique is used to identify the model of the virtual machine tool with virtual sensors. Periodic multi-sine (with frequency components between 10 and $500Hz$) excitation experiments are performed in order to estimate frequency response functions (FRFs). These excitation signals are injected as an input current to the motors. The current signals are converted to force and torque (correspond to the linear and rotary motors, respectively), for the virtual prototype. During these experiments, the position controller is disabled, whereas the velocity controller is detuned. The reference velocity is set to zero. The FRFs from X-axis motor torque to the displacements of the rotary motor and

TCP in X-direction are shown in Figures 9 and 10, respectively. In addition, the FRFs from Y-axis force to the displacements of the linear motor and TCP in Y direction are shown in Figures 11 and 12, respectively.

The following observations are made:

- **X-axis:** Both for the motor encoder and TCP FRFs, the small difference in phase around at low frequencies indicates the existence of friction in the X-axis driveline. For the encoder FRF, both the magnitude and phase correspond very well up to a certain level of accuracy throughout the frequency range of interest. However, for the TCP FRF, there are significant discrepancies for the frequencies higher than $125Hz$. The flexible mode near $185Hz$ correlates with the simulation in terms of frequency; however, the magnitude differs. The mismatches for the frequencies higher than $125Hz$ are due to the un-modelled dynamics of the scanning head bracket fixture or uncertain parameters. This is currently under investigation.

- **Y-axis:** Similar to the X-axis, both for the motor encoder and TCP FRFs, the small difference in phase at low frequencies indicates the existence of friction in the Y-axis driveline. The mass line behavior for these FRFs matches very well with the simulation model for the frequencies up to $100Hz$. For the encoder FRF, the small discrepancies at higher frequencies (such as at $150Hz$) are due to the joint stiffnesses between different components inside the head. For the TCP FRF, the discrepancies at higher frequencies, both in magnitude and phase, are might be due to the un-modelled dynamics of the scanning head bracket fixture. This can be investigated by either a numerical model or experiments.

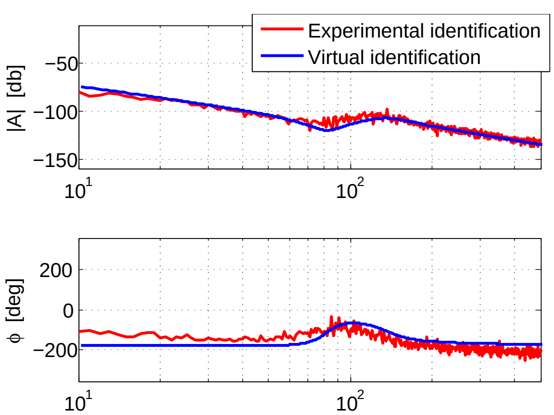

Figure 9. FRF from X motor torque to X motor encoder

An experimental validation of the numerical model shows good comparison but also a number of mismatches due to several possible reasons. These include (i) uncertain parameters (such as stiffness in the guides, damping

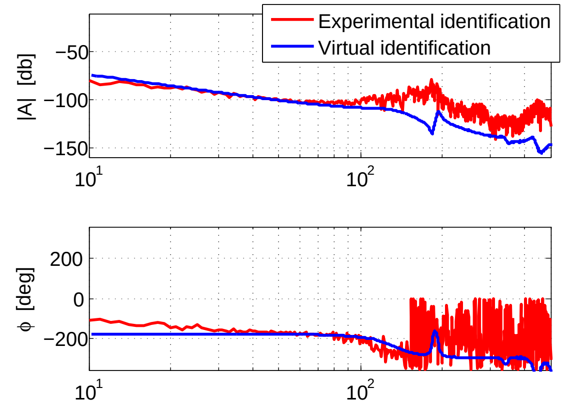

Figure 10. FRF from X motor torque to X displacement of TCP

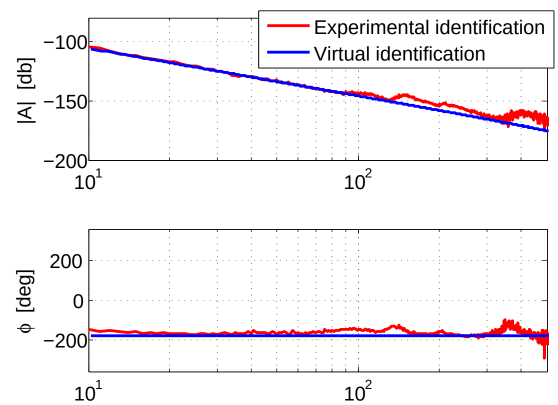

Figure 11. FRF from Y motor force to Y motor encoder

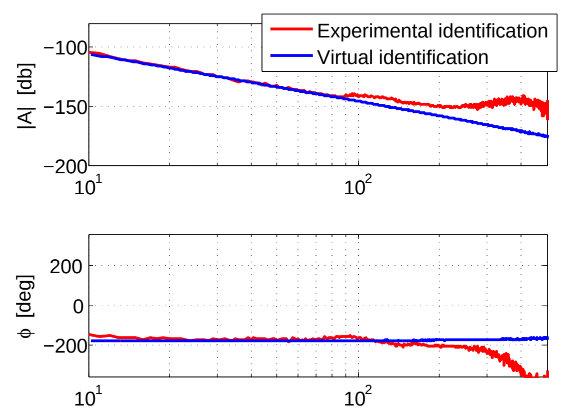

Figure 12. FRF from Y motor force to Y displacement of TCP

in the system, material properties), (ii) modeling assumptions and simplifications (for instance, rigid head assumption, neglected friction and flexibilities in the joints), (iii) manufacturing tolerances, (iv) un-modeled dynamics(for instance, fixture assembly for the sensor), (v) environmental and other boundary conditions. The stiffness of the driveline is tuned by 2% to have a better match with the experimental results. However, more experimental investigations are required to obtain an improved model. In future, the possible causes will be investegated further. Consequently, this will facilitates to tune and update the model.

4 Conclusion

In this work, the on-going activities on the simulation and validation of 1D-3D combined models for a machine tool are presented. An industrial machine is used for the demonstration of the proposed strategy. A flexible multi-body model and a 1D lumped parameter model together with the controller are developed. A co-simulation is set up between these models.

It is shown that unlike 1D lumped modeling approach, the flexible multi-body approach allows us to model the elastic deformation behavior of the system (i.e. gantry). However, in order to correlate the virtual model with reality, the modeler has to use engineering intuition, assumptions and experience, experimental data and analysis to decide on various factors; for instance, which parts of the machine can be assumed rigid. Once the model correlates well with the experiments, the model is a very useful tool to predict the dynamic behavior of the machine at early design stages, to forecast the influence of specific design changes, and to assess the impact of different control architectures. This helps to reduce the time consuming and costly procedure of making physical prototypes after every design change.

Acknowledgment

The research of S. Moten is funded by an Early Stage Researcher grant within the European Project IMESCON Marie Curie Initial Training Network (GA 264672). The IWT Flanders within the Dilacut project is also gratefully acknowledged for its support. Finally, this work also benefits from the KU Leuven Optimization in Engineering Center (OPTEC) and Belgian Programme on Interuniversity Attraction Poles, initiated by the Belgian Federal Science Policy Office (DYSCO). Goele Pipeleers is Postdoctoral Fellow of the Research Foundation Flanders (FWO).

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